

Vibration platform testbed for deep space acquisition, tracking and pointing

Juan M. Cenicerros, Christian D. Jeppesen, Gerardo G. Ortiz

Jet Propulsion Laboratory, California Institute of Technology
M/S 161-135, 4800 Oak Grove Drive, Pasadena, CA 91109

ABSTRACT

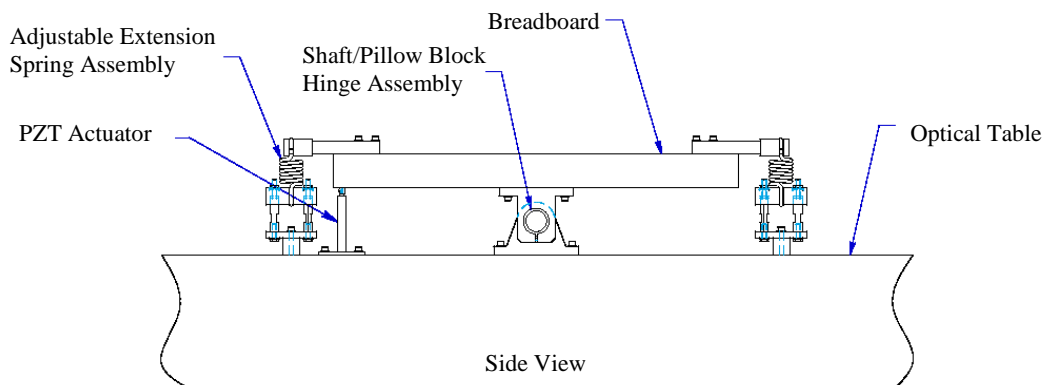
Precise Acquisition, Tracking and Pointing (ATP) remain a key issue in the use of free-space optical communication systems for deep space missions. The Optical Communications Group at the Jet Propulsion Laboratory is developing a vibration platform to assist with the development and characterization of an ATP system to be used for deep space optical communications. The vibration platform will provide a means for subjecting ATP systems to the vibration spectrum likely to be experienced while onboard a spacecraft. The platform consists of a 61 cm x 61 cm optical breadboard mounted on a ball bearing pivot that is driven by a single piezo-electric actuator (PZT). The PZT provides motion in a single axis, giving the platform approximately 200 μ rad of angular motion with a bandwidth in excess of 100 Hz. When placed on the platform, the performance of ATP systems can be tested under several cases of vibration. This paper will discuss the physical properties of the vibration platform. A model for the system will be discussed and experimental performance data will be presented.

1. INTRODUCTION

The Vibration Platform Testbed (VPT) is being developed as a low cost, single axis vibration platform to assist with the development and characterization of acquisition, tracking and pointing (ATP) systems for deep space optical communication at JPL. To accurately point at the earth, an optical communication system for a deep space mission must possess the capability to correct for jitter induced by the spacecraft the system is onboard. Ideally, the VPT will subject the optical communication system under test to a vibration spectrum that is likely to be encountered while onboard a spacecraft. The goal of the VPT is to provide single axis motion with a vibration spectrum out to 150 Hz and a maximum angular displacement of 100 μ rad.

2. VPT SYSTEM

The VPT system consists of a 61 cm by 61 cm optical breadboard, mounted on a shaft/pillow block assembly that is used as a pivot, and driven by a piezo-electric actuator (PZT). The breadboard is a Melles Griot UltraPerformance Series (product number 07 OBC 005) chosen for the high compliance specifications. The shaft/pillow block assembly is mounted directly under the centerline of the breadboard. The PZT, manufactured by Polytec PI (model P-841.40), is located below one end of the breadboard and provides the motion for the system. As the PZT should only operate in either a push or a pull mode, but not both, it is highly advisable not to mount the breadboard directly to the PZT. A "ball bearing tip" was manufactured to serve as the contact point between the PZT and the breadboard. Since it was not possible to directly mount the breadboard to the PZT, one expansion spring is located at each corner to provide the force necessary to keep the breadboard in contact with the PZT. Sketches of the system are shown in Figure 1.



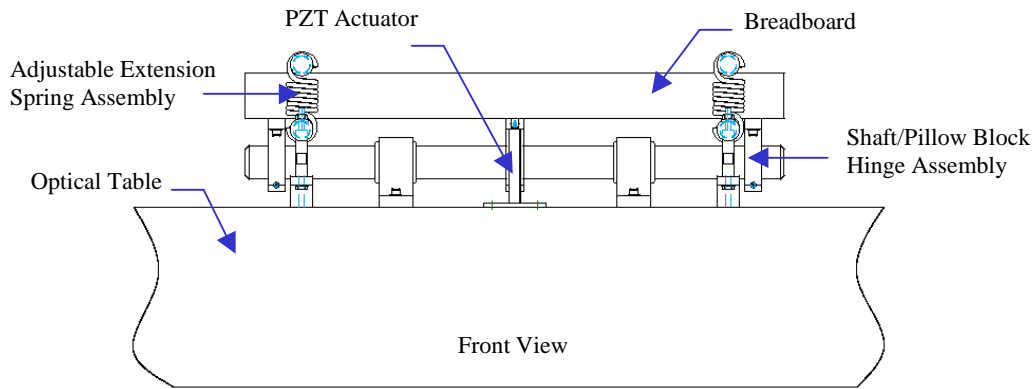


Figure 1. Sketches of VPT.

Whereas several manufacturers produce three axis motion systems, the VPT is unique in the sense that it provides single axis motion. This feature was highly desirable as the ATP system at JPL is under development and testing the system in one axis before evolving to three axes seems logical. The surface of the VPT is an optical breadboard, no further modifications are necessary to mount optics. Another feature is that the motion of the PZT actuator can be observed and several actuators can be interchanged with others to provide various ranges of displacement, resolutions, bandwidths, and handle various loads. The family of actuators supported by the driver can provide maximum angular displacements from $50 \mu\text{rad}$ to $300 \mu\text{rad}$. The resolution for these devices range from $9.84 \times 10^{-4} \mu\text{rad}$, for the smaller displacement, to $5.91 \times 10^{-3} \mu\text{rad}$ for the larger displacement. The various actuators can also provide pushing forces from 800 N to 3000 N. The 3 dB bandwidth of these devices ranges from 30 Hz to 700 Hz. The waveform used to drive the actuator can be a computer generated file or come from a signal generator and can be of any form, it is not limited to simple sine or square waves. Furthermore, the springs used to balance the system can compensate for uneven mass distribution placed on the optical breadboard, allowing the optical path to take on several forms.

3. PIEZO-ELECTRIC ACTUATOR

The PZT actuator used is the Polytec PI model P-841.40. It provides $60 \mu\text{m}$ of displacement with a pushing force of 1000 N. The driver used with the PZT is the E-662.SR, also manufactured by Polytec PI. With the P-841.40 as the load, the driver has a 3 dB bandwidth of approximately 150 Hz. The driver can provide maximum displacement out to roughly 90 Hz [1].

The PZT was characterized while completely unloaded. A random number generator was used to create a white noise spectrum that provided the input to the PZT driver. The actual motion of the PZT was measured by observing the output of the strain gauge position sensor integrated with the PZT. The noise spectrum used as the input to the PZT driver caused the PZT to displace a total of $15 \mu\text{m}$, well above the noise floor of the PZT position sensor. A LabView program was created to both send the control signal the driver and read the output of the position sensor. The update rate of the program was 5 kHz, however a 1 kHz filter was used for the control signal provided to the PZT driver. The transfer function is shown in Figure 2. Small peaks can be seen in the PZT response at approximately 100 Hz and 190 Hz.

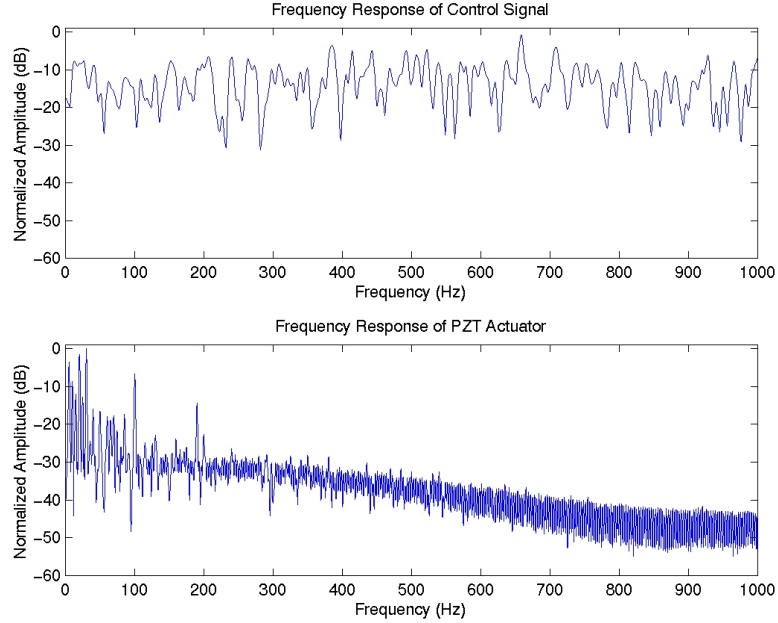


Figure 2. Transfer function of random control signal and PZT actuator.

A model for the piezo displacement was derived using Matlab. State-space models were estimated with orders from 2 to 10. The model that produced the best fit was a N4S2 second order model. The equation for the transfer function that describes the model is given by

$$H(z) = \frac{1.0374z + 0.0231}{z^2 - 0.9262z - 0.0288}$$

The frequency response of the model is shown in Figure 3. The model appears to follow the response of the PZT actuator fairly well. The major difference is that the model does not appear to drop off as quickly at low frequencies.

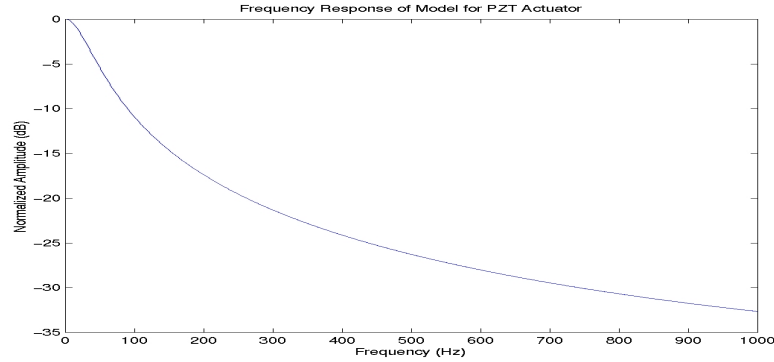


Figure 3. Frequency response of model derived for PZT actuator using Matlab.

4. VPT PERFORMANCE

The performance of the VPT was characterized using a position sensitive detector (PSD), which computes the centroid of a spot that strikes its active area. The PSD used is the Model OT-324.0 from On-Trak Photonics with an active area of 4 mm by 4mm and a typical resolution of 500 nm. The PSD was placed on the breadboard directly above the PZT. The reference beam was provided by a red He-Ne laser, and was located on a separate optical bench than the VPT, to provide a higher level of isolation from the motion of the VPT. Figure 4 shows the response of the PSD for a static platform. The PSD detected a peak to peak motion of 888.99 nm whereas the sensor for the PZT actuator detected a peak to peak motion of 59.99 nm. A second test was performed to check the measurement error of the PSD by mounting the PSD rigidly to the PZT. The measurement error was approximately 15% for a 25 Hz sine wave with peak to peak motion of 24 μm .

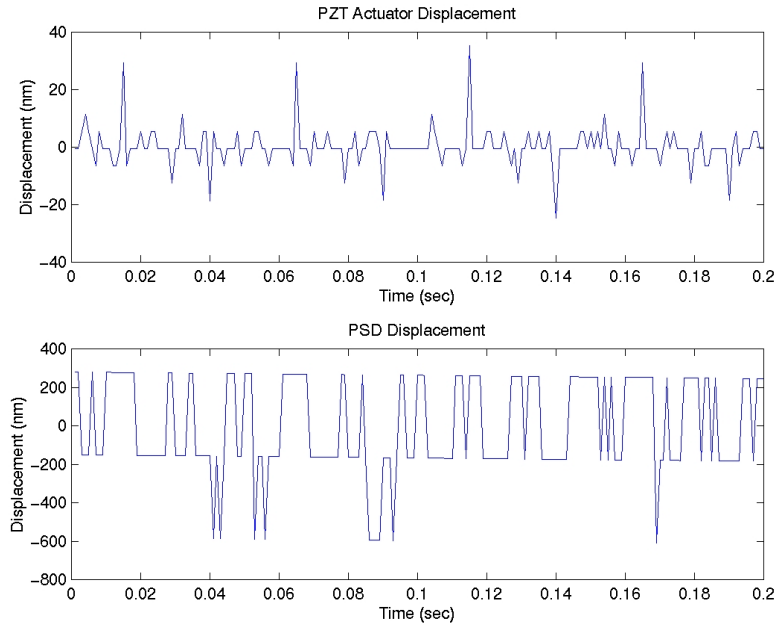


Figure 4. Static performance of PZT and PSD when used on the VPT.

The same random sequence that was used to characterize the performance of the PZT was again used to characterize the VPT performance. A LabView program was used to sample at the PSD output 5 kHz and the same 1 kHz filter was used for the outgoing control signal provided to the PZT driver. The control signal causes the PZT actuator to displace approximately 12-15 μm , which is well above the noise floor of the PSD. The transfer function of the VPT system is shown in Figure 5. The response of the VPT has a large resonance at 67 Hz. This resonant frequency can be slightly adjusted, from 65 Hz to 75 Hz, depending on the amount of tension placed on the springs. A second resonance is clearly visible at approximately 185 Hz.

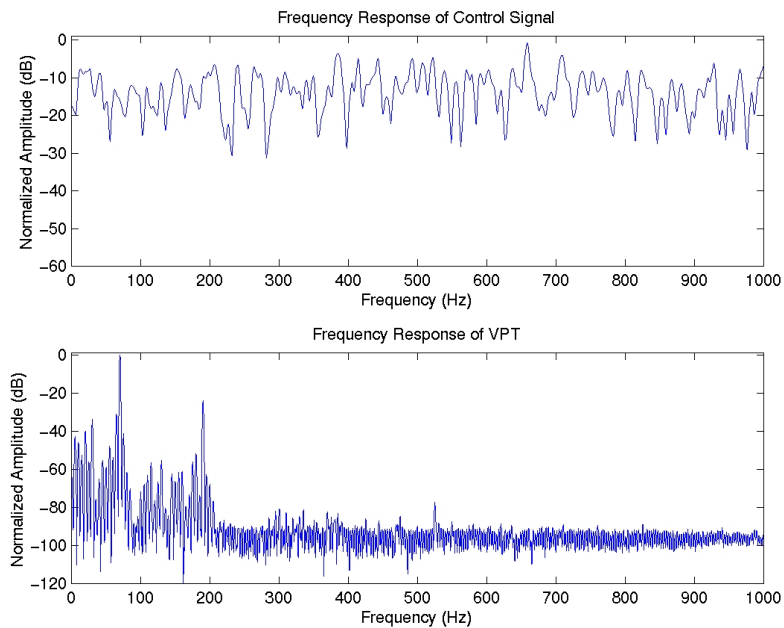


Figure 5. Frequency response of Vibration Platform Testbed.

A model for the VPT displacement was also derived using Matlab. Again state-space models of order 2-10 were generated to determine which order possessed the best fit. As was the case for the PZT, a third order N4S3 model was optimal. The equation for the transfer function of the model is given by

$$H(z) = \frac{0.0047z^2 - 0.0013z + 0.0062}{z^3 - 0.8448z^2 + 0.0374z - 0.0785}$$

The frequency response of the model is shown in Figure 6. This time the model does not precisely match the response of the VPT. The large peaks at 67 and 185 Hz do not show up on the response of the model, also the VPT has a fairly flat response past 200 Hz, whereas the model continues to show a decrease.

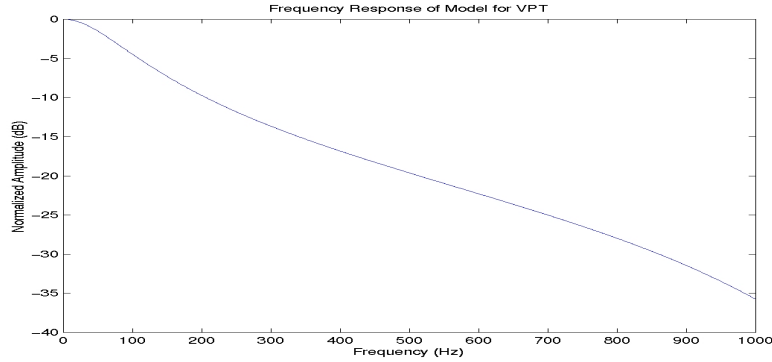


Figure 6. Frequency response of model derived for VPT using Matlab.

Another characteristic of the VPT that was measured was the relative motion from one portion of the breadboard to another. This was performed by moving the PSD to each of the four corners of the breadboard as well as from the front to the back of the center line (where the PZT is located). Figure 7 shows the locations where the PSD was placed. The PZT is located underneath position 4 in the figure and the shaft/pillow block assembly runs along the line connecting positions 2, 5 and 8.

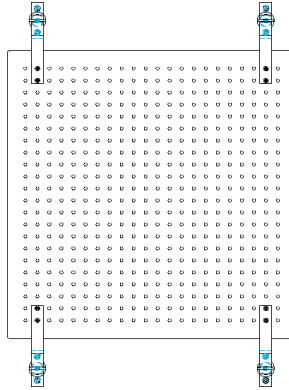


Figure 7. Locations on VPT where PSD was placed to measure displacement.

The VPT was excited with three sinusoids of various frequencies, 25 Hz, 50 Hz and 100 Hz. Each sinusoid caused the PZT actuator to displace a total of 12-21 μm . The PSD measured the displacement at each of the locations shown on Figure 7. If the breadboard were an ideal rigid body, the displacement at positions 1,3,4,6,7 and 9 would be identical to the displacement of the actuator and positions 2, 5 and 8 would show no motion. Laboratory experiments show that this is not the case. The actuator was excited with sinusoidal signals and the displacement was measured with the PSD. Sinusoids of 25 Hz, 50 Hz and 100 Hz were used for this characterization. The motion detected by the PSD when place at position 1 when the VPT is actuated with a 25 Hz signal is seen in Figure 8. The figure also shows the frequency response of the PSD signal. The signal from the PSD is fairly clean and follows the 25 Hz drive signal very well. From the frequency response it is clearly seen that 25 Hz is the dominant signal. However, small content around 50 Hz, 60 Hz and 70Hz is also apparent.

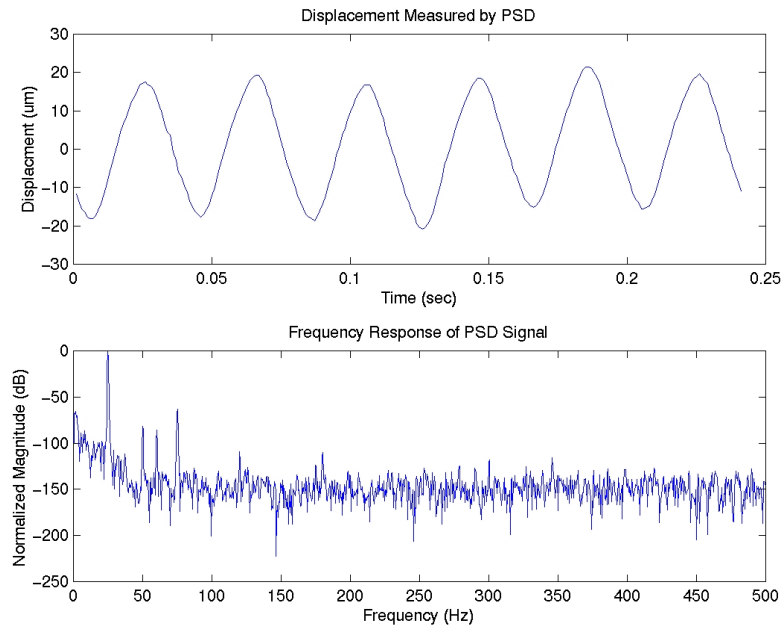


Figure 8. Signal seen by PSD in position 1 when excited with 25 Hz sinusoid.

The response obtained in position 5 when driven by a 50 Hz signal is seen in Figure 9. As position 5 is located directly above the pivot, the detector should ideally measure no change in position. The PSD does detect motion, however it is clearly smaller than the motion seen by the locations away from the pivot. From the frequency response we can see that DC terms dominate the signal content, and not the 50 Hz that the actuator is moving at. Thus the locations above the pivot move less than those away from the pivot, but do not behave in an ideal fashion.

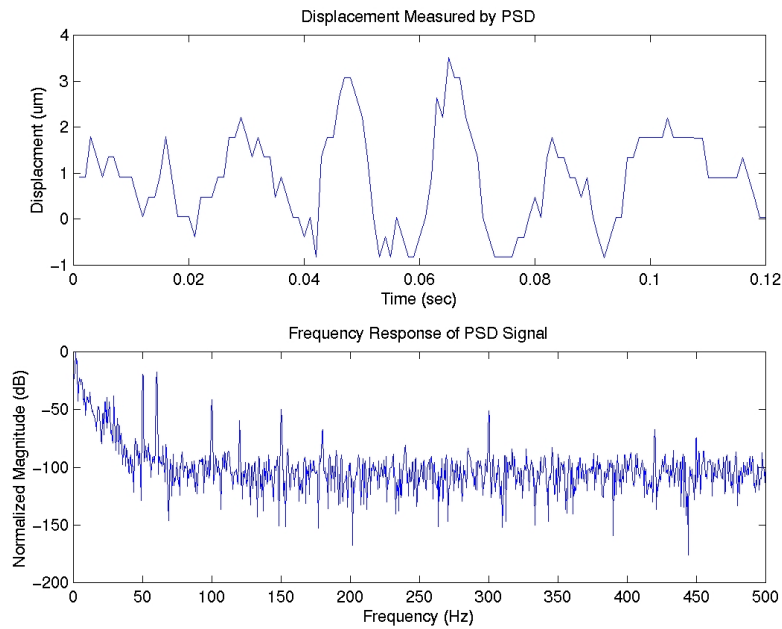


Figure 9. Signal seen by PSD in position 5 when excited with 50 Hz sinusoid.

The response of the PSD when placed location 9 and driven by a 100 Hz signal is shown in Figure 10. The PSD signal clearly follows the 100 Hz sine wave. From the frequency response we can see that the majority of the signal content is located at 100 Hz, with other significant contributions at DC and approximately 65 Hz.

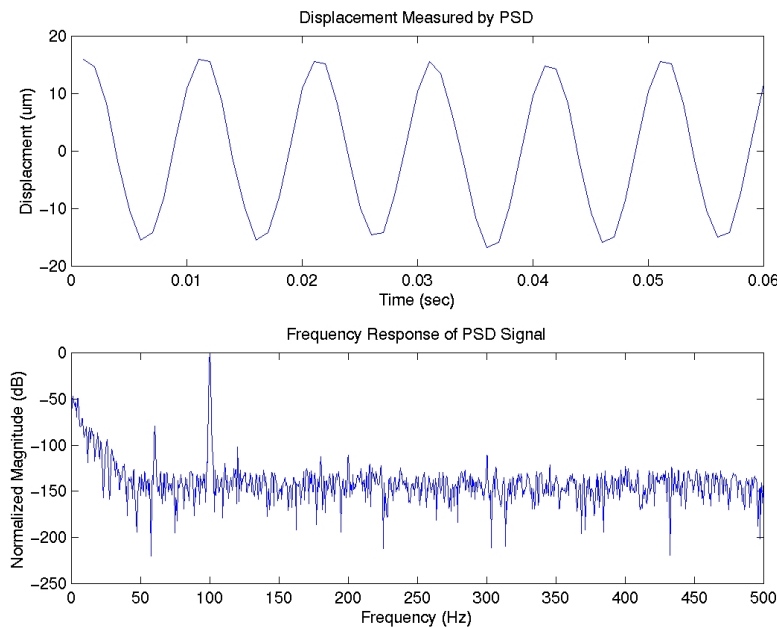


Figure 10. Signal seen by PSD in position 9 when excited with 100 Hz sinusoid.

It is easily seen that the breadboard does not behave as an ideal rigid body. There is no simple solution to solve the problem, as the breadboard weighs approximately 43 kg and a more rigid platform that would provide better response would certainly increase the amount of weight the PZT actuator must displace. The platform does, however, behave relatively well and should perform adequately.

5. VIBRATION SIMULATION WITH VPT

A file was created to simulate a vibration spectrum. This file contains frequency content from approximately 10 Hz to 260 Hz. The content at the lower frequencies is dominant and the decay is second order. The frequency response of this control signal is seen in Figure 11. Also seen in Figure 11 is the response of the PZT actuator to this drive signal. It is clearly evident that the actuator can not keep up with control signal. The roll off is much sharper and a noticeable local minimum occurs at 70 Hz. Still, the actuator does provide motion out to 260 Hz and should be suitable as the drive mechanism for the VPT.

The PSD was placed at position 4 to measure the amount of displacement when the VPT was driven by the vibration simulation file. The frequency response of the PSD signal is also seen in Figure 11. From this response is clearly evident that the majority of the signal content is located around 65 Hz, which is the resonant frequency of the VPT system. Another resonant mode is also apparent at 180 Hz. Just as the actuator response showed a local minimum around 70, the same is true for the VPT. Unlike the actuator, however, the response of the VPT actually increases from 100 Hz to 200 Hz. There is a sharp decay from 200 Hz to 260 Hz and the response is flat beyond that point. Whereas the performance of the VPT is far less than ideal, the VPT does possess motion out to approximately 200 Hz. As such, it would seem possible to produce a digital filter that would condition the PZT control signal such that the desired vibration spectrum is achieved.

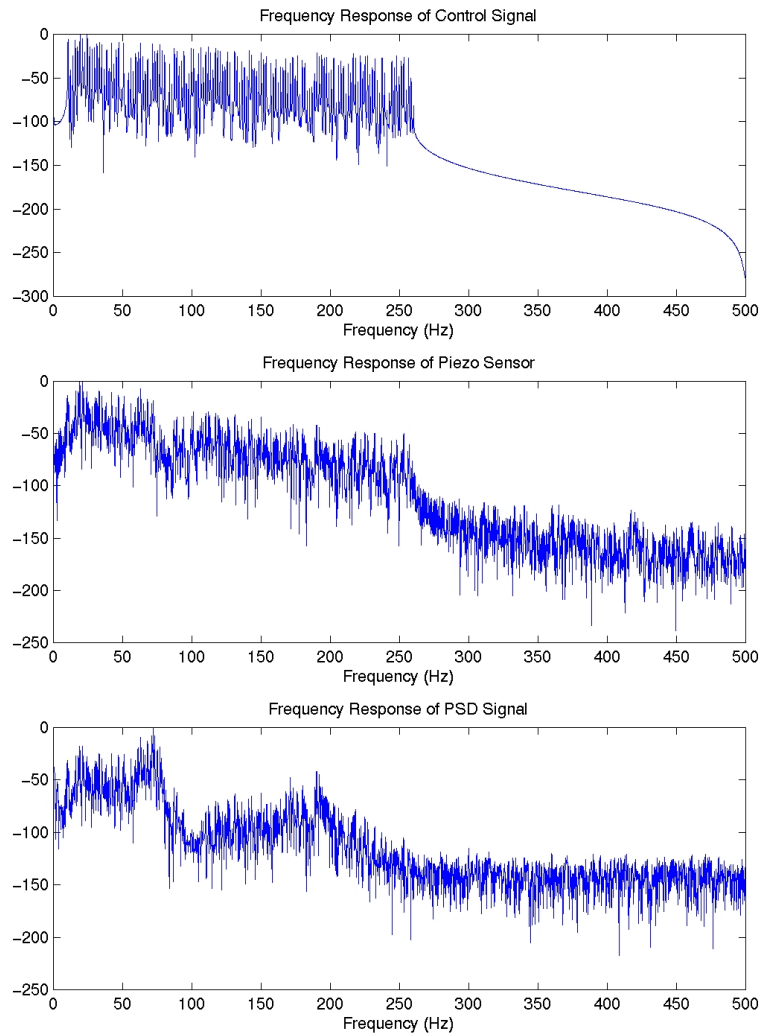


Figure 11. Frequency response of vibration simulation file and PZT actuator.

6. OPTIC MOUNTS

Due to the vibration the optics on the VPT will be exposed to, it is necessary to have rigid mounts for all optical components to be used. The beam path is approximately 5 cm off the top of the breadboard used for the VPT. This is because the largest lens necessary has a 7.62 cm diameter. Standard posts and holders are undesirable as the standard thread by which they hold mounts could easily become undone causing the optics to rotate or jitter. The posts could also begin to flex when exposed to vibration.

Custom mounts were made for each of the optical components to be used on the VPT. A series of “V” brackets were made to hold all the parts of the acquisition, tracking and pointing setup, consisting of lenses, a camera and a fine steering mirror. Each “V” bracket resembles an upside-down “V” and is made of aluminum. The main wall of each bracket (where the component is threaded into the bracket) is 6.35 mm thick and all other sides are 3.175 mm thick. Optical cell assemblies from Edmund Scientific are used to hold the lenses that are used in the VPT setup. The cell assembly is then threaded into a “V” bracket. A set screw is used to keep the thread from breaking loose while under vibration. Figure 12 shows a diagram of an optical cell assembly mounted on a “V” bracket.

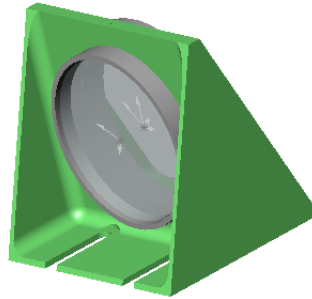


Figure 12. Optical cell assembly mounted to “V” bracket.

The performance of the mounts remains to be tested. It is fully expected that each mount will provide a firm interface for the component it holds such that the components will not separate itself from the mount. Furthermore, these mounts are extremely rigid and should not have a natural frequency under 200 Hz. The performance of mounts is expected to be sufficient for use on the VPT.

7. CONCLUSIONS

The VPT is a low cost assembly that can be used to characterize and test the performance of the Acquisition, Tracking and Pointing system developed by the Optical Communications Group at JPL. The VPT contains a 61 cm by 61 cm breadboard on a pivot to obtain single axis motion. The VPT assembly is shown in Figure 13 and Figure 14. The CCD camera is seen at the upper portion of the figure. The fine steering mirror used to point the beam is located directly across from the CCD and accelerometer is seen towards the front of the table. Several optics used to focus the beam and split the paths are also seen. Figure 14 provides a better view of the fine steering mirror, and the transmitted beam can be seen at the center of the reflective surface. The displacement is provided by a PZT actuator which can provide motion out to approximately 150 Hz with a 3 dB bandwidth. The maximum angular displacement at this frequency is approximately 100 μ rad. At lower frequencies the VPT can provide 200 μ rad of angular displacement. By interchanging the PZT actuator with another model, it is possible to obtain a larger range of motion, higher resolution, higher bandwidth, or greater load handling capability.

As the optical breadboard used for the VPT is not a perfectly rigid structure, each portion of the platform does not possess a one-to-one relation with the motion of the PZT actuator. The displacement seen by the VPT when the actuator displaces 200 μ rad can be as little as 40 μ rad or as large as 270 μ rad, depending on what position of the VPT is used. The VPT also contains resonance frequencies at approximately 67 Hz and 185 Hz. By varying the amount of tension in the springs used to balance the VPT, the resonant frequency can be slightly adjusted and dampened. These resonant frequencies make it difficult to accurately model the system using a state-space approach. The VPT does not possess the ability to be modified to provide three axis motion. A new system would need to be procured or developed.

ACKNOWLEDGEMENTS

The work described was funded by the Cross-Enterprise Technology Development Program and performed at the Jet Propulsion Laboratory, California Institute of Technology under contract with the National Aeronautics and Space Administration. The authors wish to thank Tim Newby for all his efforts in designing and fabricating the specialized interface between the optical breadboard and the shaft/pillow block assembly, the interface between the breadboard and the PZT actuator, and the hardware to mount the springs onto the breadboard and optical bench. The authors also want to thank Jerry Eden for designing and fabricating the “V” brackets.

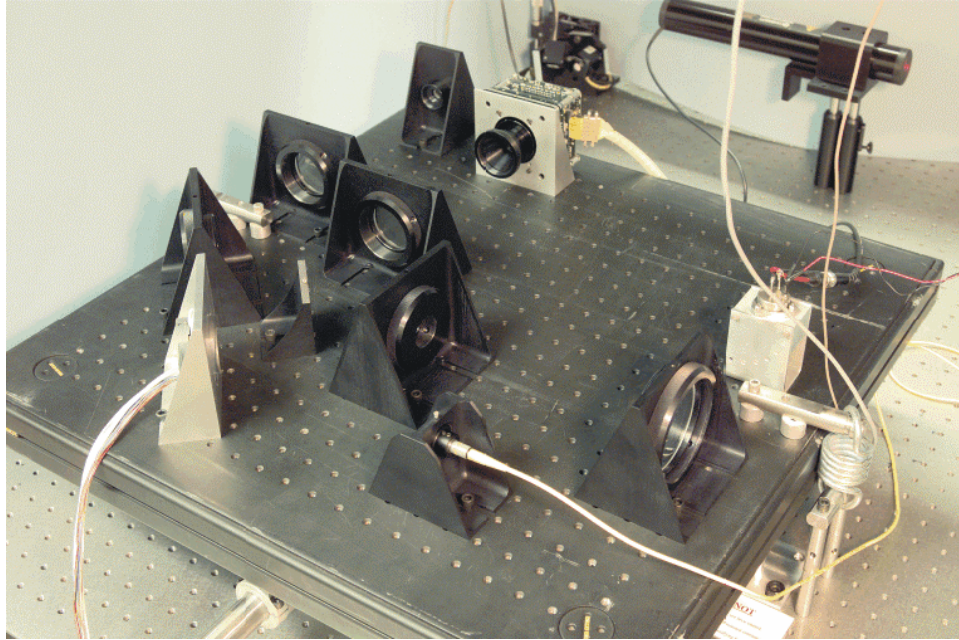


Figure 13. Vibration Platform Testbed Assembly.

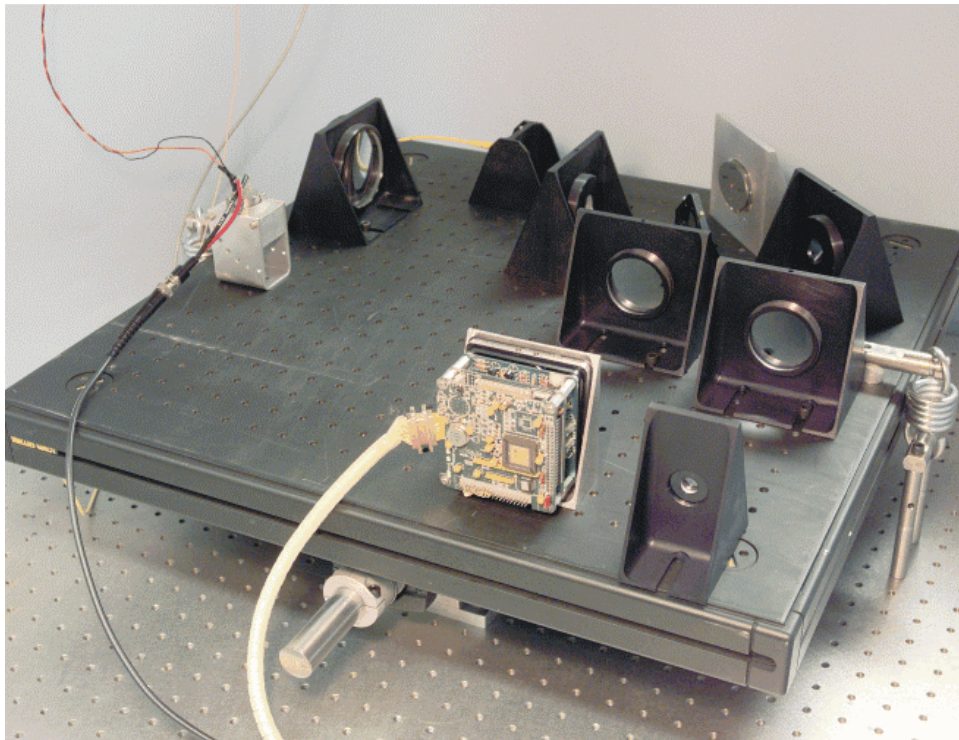


Figure 14. Vibration Platform Testbed Assembly.

REFERENCES

1. Physik Intrumente NanoPositioning Catalog 1998.